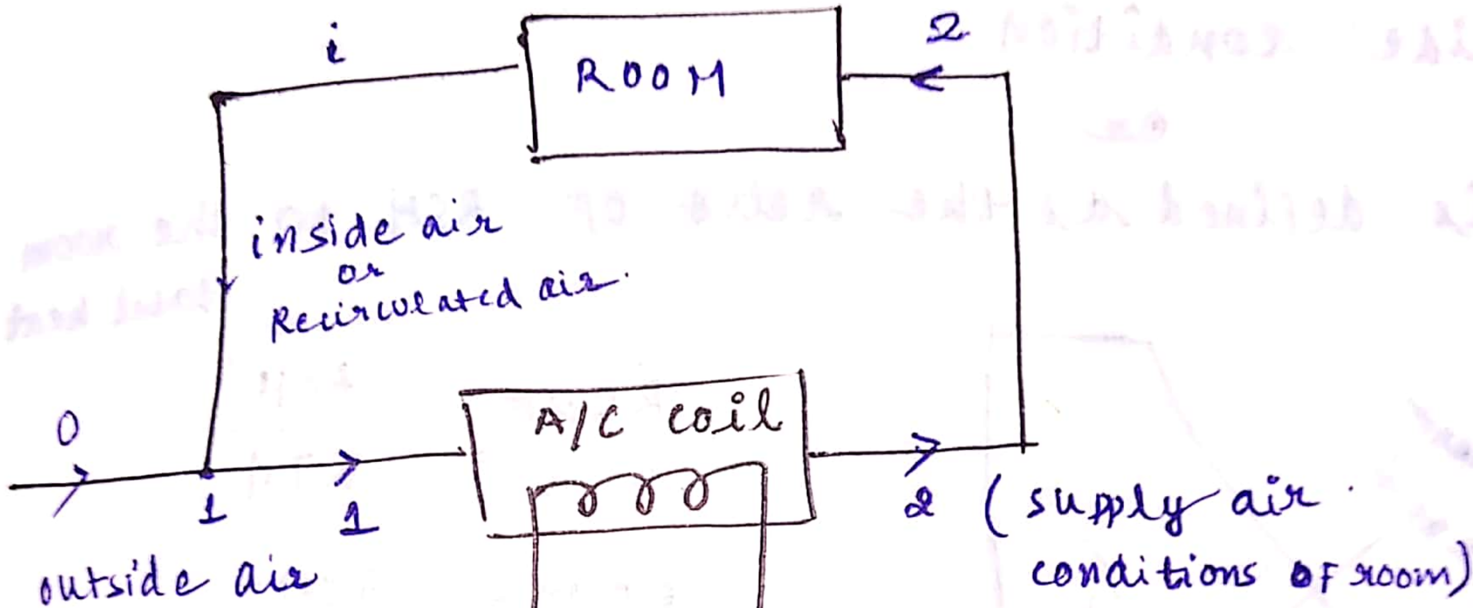
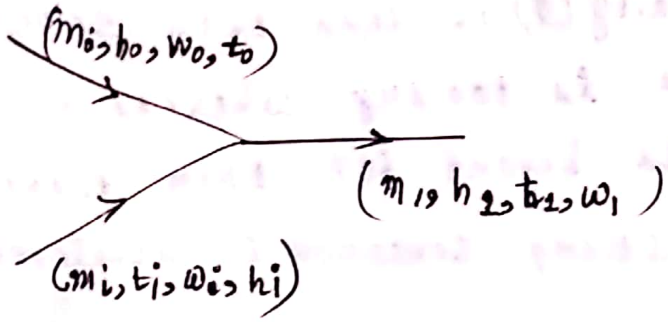
CASE-I Summer A/c with zero By-pass Factor



Summer A/c
cooling + Dehumidification.
(x = 0)

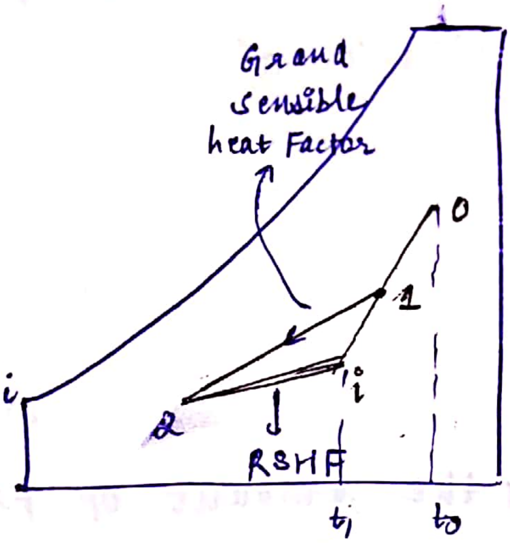


$$m_1 = m_o + m_i$$

$$m_1 h_2 = m_o h_o + m_i h_i$$

$$m_1 t_1 = m_o t_o + m_i t_i$$

$$m_1 w_1 = m_o w_o + m_i w_i$$



GSHF:

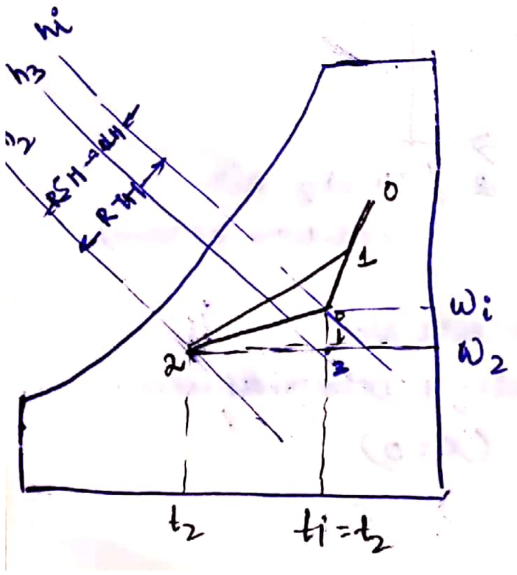
It is the line obtained by joining the inlet & outlet cond's of the a/c coil

RSHF (Room sensible Heat Factor)

It is the line obtained by joining the supply cond" of the room with the inside condition.

Or

It is defined as the ratio of RSH to the room total heat



$$RSHF = \frac{RSH}{RTH}$$

$$RSHF = \frac{RSH}{RSH + RLH}$$

$$\boxed{RSHF = \left(\frac{h_3 - h_2}{h_i - h_2}\right)}$$

NOTE

1) RSH = 0.0204 Cmm Δt (KW)

2) RLH = 50 cmm Δw (KW)

$\Delta w = w_i - w_2$

Cmm = m³/min

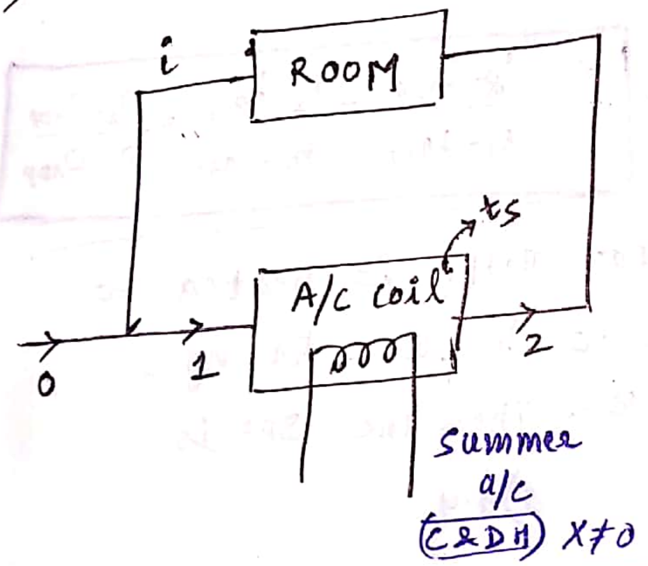
Cmm → volume flow rate of air taken in m³/min.

3) No. of air flow changes/hr = $\frac{\text{cmm}}{\text{vol. of room}(m^3)}$

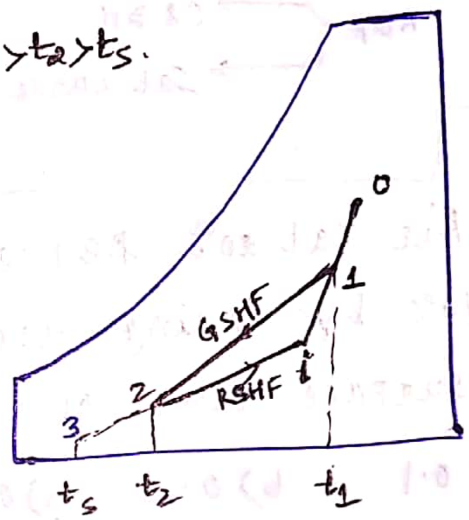
cmm → m³/hr

CASE-II

Summer A/c with non-zero By-pass Factor

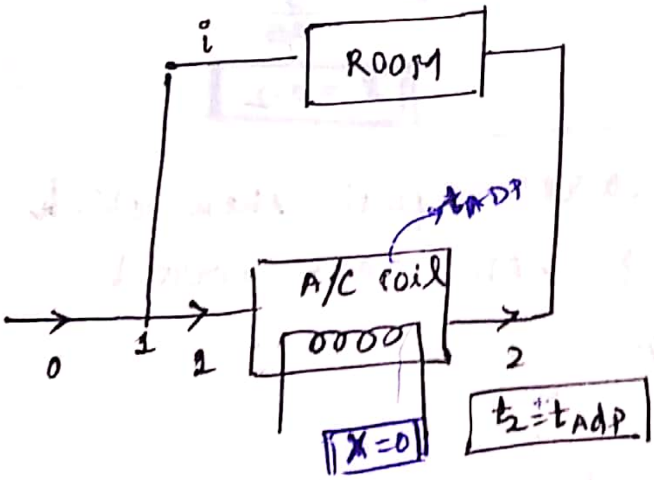


$t_1 > t_2 > t_s$.

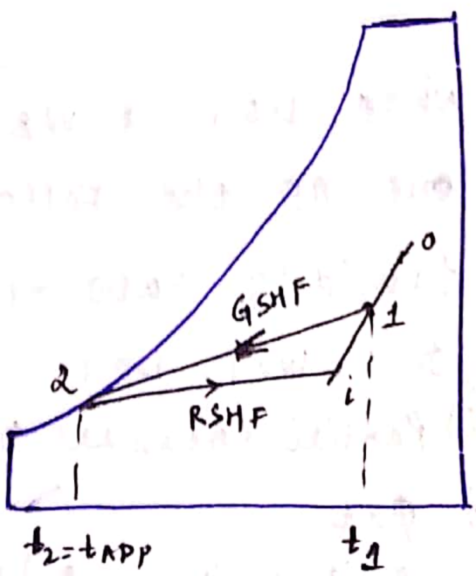


CASE-III

Air passing thru a coil having some given value of ADP with zero Bypass Factor

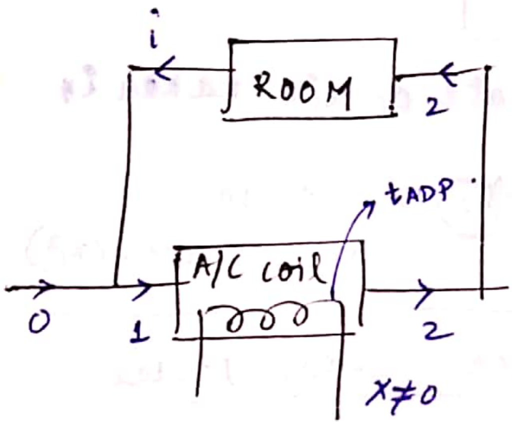


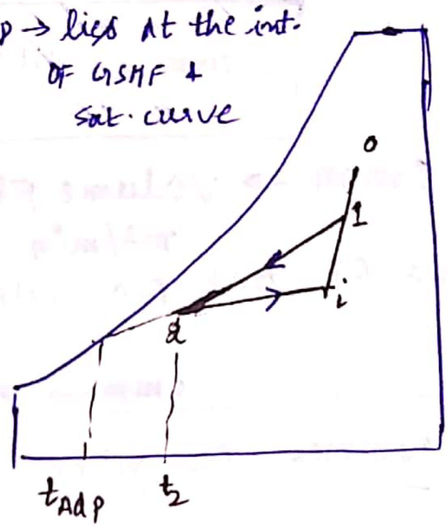
ADP → C & DH
ADP → Sat. curve



CASE-IV

Air passing thru a coil having some given value of ADP with non-zero by pass factor.





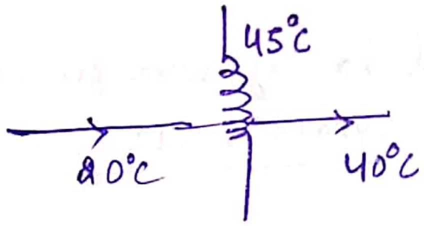
ADP → C & DH
ADP → sat. curve.

$$x = \frac{t_2 - t_{ADP}}{t_1 - t_{ADP}} = \frac{h_2 - h_{ADP}}{h_1 - h_{adp}} = \frac{\omega_2 - \omega_{ADP}}{\omega_1 - \omega_{ADP}}$$

IES:

Q) Air at 20°C DBT & 40% R.H. is heated to 40°C by using electric heater having a surface temp of 45°C. Then the BPF is

a) 0.1 b) 0.2 c) 0.3 d) 0.4



$$x = \frac{45-40}{45-20}$$

$$x = \frac{5}{25}$$

$$\boxed{x = 0.2}$$

Q) When DBT & WBT are equal then which one of the following comb's are correct.

i) Humidity ratio = 100%.
ii) DBT = WBT = DPT
iii) Partial pressure of water vapour = Total pressure
iv) $\phi = 1$

A) 1, 2, 4 B) 2, 4 ✓c) 2, 3, 4 d) 1, 3

Q) The BPF of a single cooling coil in a A/C is 0.7, then the By pass of three such cooling coils with same surface temp. one behind the other.

$(0.7)^3 = 0.343$

Q) (IES) For an air conditioned space Room total heat is 100 KW. RSHF = 0.75, volume flow rate = 100 m3/min. Indoor sp. humidity = 0.01 Kg/kg of d.a.. then the sp. humidity of supplied air is :-

a) 0.01 b) 0.0075 c) 0.005 d) 0.015.

$RSHF = \frac{RSH}{RTH} \Rightarrow 0.75 = \frac{RSH}{100}$

$\Rightarrow$ **RSH = 75 KW**

$RSH = 0.0204\,(cmm) \times \Delta t$ **RLH = 25 KW**

$75 = 0.0204 \times 100 \times \Delta t$

$\Rightarrow \Delta t = \frac{75}{2.04} = 37.5$.

~~$\omega_i = 0.01$~~

$R.LH = 50 \times (cmm) \times \Delta W$

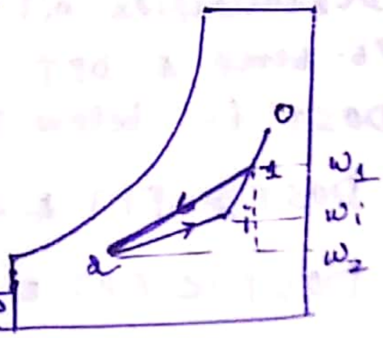
$25 = 50 \times 100 \times \Delta\omega$

$\Delta\omega = \frac{1}{200}$

$(\omega_i - \omega_2) = 0.005$.

ω_i: $0.01 - \omega_2 = 0.005$

$\boxed{\omega_2 = 0.005}$

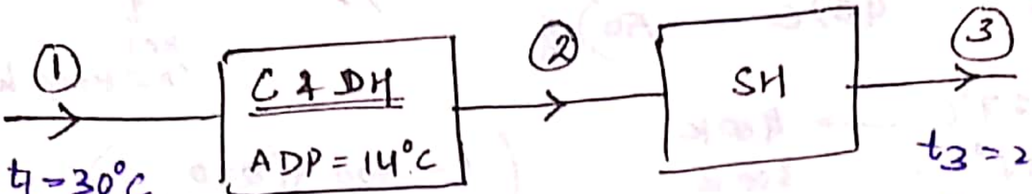


Conventional Questions

Q) IES-92
A small auditorium is req. to be maintained at 22°C DBT & 70% RH and the ambient cond's are 30°C DBT & 75% RH. The amt. of Free air circulated is 200 m³/min. The reqd cond's are achieved by First cooling & dehumidification thru a cooling coil having an apparatus dew pt. of 14°C and then by heating with the help of Psychrometric chart Find

i) cooling capacity of the cooling coil in Tonnes of Ref. and its by pass Factor.

ii) amount of water vapour removed by the cooling coil in Kg/hr.



$t_1 = 30°C$ $\quad$ $t_3 = 22°C$

$\phi = 75\%$ $\quad$ $\phi = 70\%$

$h_1 = 82$ KJ/Kg

$\omega_1 = 0.020$ Kg/Kg of d.a

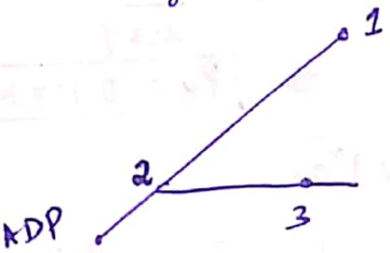
$v_1 = 0.83$ m³/Kg

$\dot{m}_1 = \frac{\dot{V}}{v_1} = \frac{200 \text{ m}^3/\text{min}}{0.88 \text{ m}^3/\text{Kg}} = \frac{200/60}{0.88}$

$\boxed{\dot{m}_1 = 3.78 \text{ Kg/s}}$

$h_3 = 51$ KJ/Kg, $\omega_3 = 0.011$, $v_3 = 0.85$ m³/kg, $h_{ADP} = 39$ KJ/kg

$\omega_{ADP} = 0.010$ kg/kg of d.a



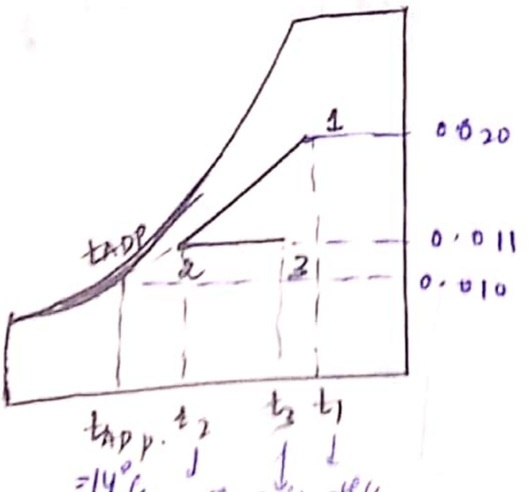
$h_2 = 46$

$\omega_2 = \omega_3 = 0.011$

STEP-I :- First locate the outside and inside points i.e state 1 & state 3 on the Psychrometric chart

STEP-II :- Locate the ADP point with the help of dry bulb temperature.

STEP-III :- The point 2 is obtained by the intersection of cooling and dehumidification line with the sensible heating



1) CC of coil $= (h_1 - h_2) \times \dot{m}$

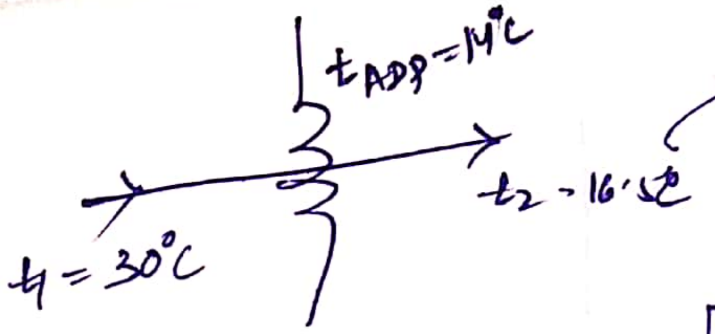
$= 3.78 \times (82 - 46)$

$\boxed{C.C = \frac{136.30}{3.5} \approx 39 \text{ TR}}$

$\dot{m} = \frac{\text{Vol}}{v_1} = \frac{200/60}{0.88}$

v_1 → entering

$= 3.78$ kg/s.



$$BPF = \frac{16.5 - 14}{30 - 14}$$

$$\boxed{BPF = 0.156}$$

ii) $(\omega_1 - \omega_2) \times \dot{m}$

$= (0.020 - 0.011) \times 3.78 \times 3600$

$= 122.47 \ kg/hr$

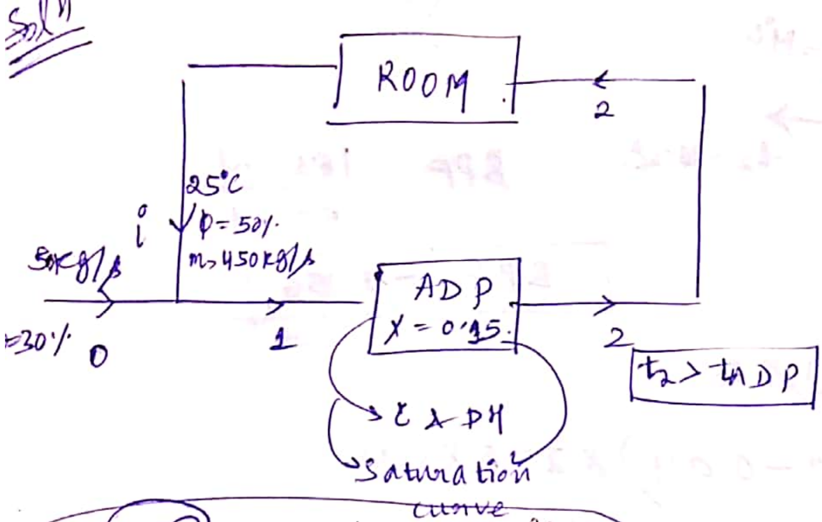
---

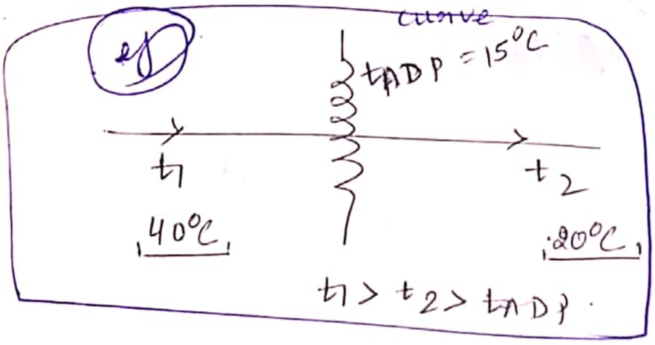
conventional ques

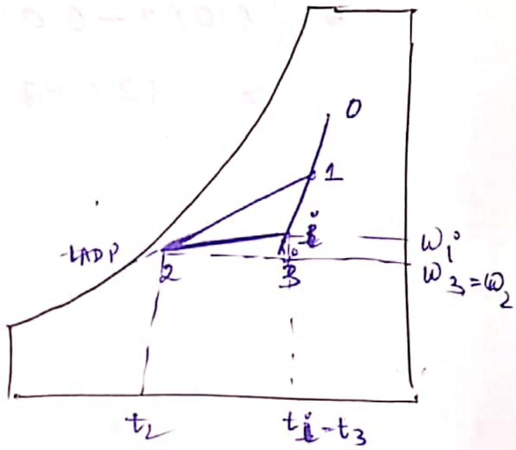
1) An air condition space 5 kg/s of outside air at 45°C at 30°C relative humidity. The room air at 25°C DBT $\phi = 50\%$ is recirculated at 450 kg/s. The mixed air flow over the cooling coil having an ADP of 12°C & 15% BPF.

Determine the condⁿ at outlet of C.C. RSH, RLH, cooling rate & condensate rate.

Sol$^{(i)}$







NOTE → ADP means $\phi = 100\%$

$h_0 = 1.005\, t_0 + \omega_0 (2500 + 1.88\, t)$

$h_0 = 1.005 \times 45 + \omega_0 (2500 + 1.88 \times 45)$

$\omega_0 = \dfrac{0.622\, P_{v0}}{P - P_{v0}} = \dfrac{0.622\, P_{v0}}{1.01325 - P_{v0}}$ ⇒ $\omega_0 = 0.018$ kg/kg of da

⇒ $h_0 = 92.17$ KJ/kg

$\phi_0 = \dfrac{P_{v0}}{P_{vs}|_{45°C}}$ ⇒ $0.3 = \dfrac{P_{v0}}{0.09584}$ ⇒ $P_{v0} = 0.0287$ bar

$\omega_i = \dfrac{0.622\, P_{vi}}{P - P_{vi}}$

$\phi_i = \dfrac{P_{vi}}{P_{vsi}|_{25°C}}$ ⇒ $0.05 = \dfrac{P_{vi}}{0.3166}$

$\omega_i = \dfrac{0.622 \times 0.01583}{P - P_{vi}}$

⇒ $P_{vi} = 0.01583$ bar

$\omega_i = 0.00987$ kg/kg of da

$h_i = 1.005\, t_i + \omega_i (2500 + 1.88\, t_i)$

$h_i = 1.005 \times 25 + 0.0987 (2500 + 1.88 \times 25)$

$\Rightarrow$ $h_i = 50.26$ kJ/kg

$h_{ADP} = 1.005 t_A + \omega_A (2500 + 1.88 t_A)$

$= 1.005 \times 12 + \omega_A (2500 + 1.88 \times 12)$

$h_{ADP} = 34.06$ kJ/kg

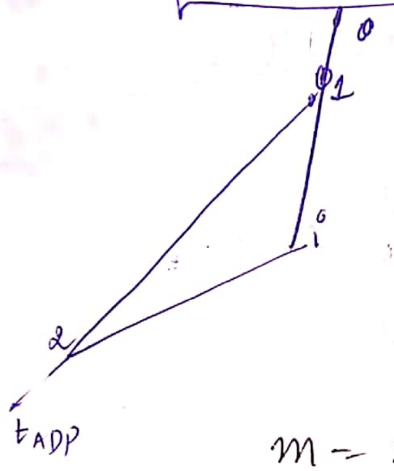
$\omega_A = \dfrac{0.622 P_{VA}}{P - P_{VA}} = \dfrac{0.622 P_{VA}}{1.01325 - P_{VA}}$

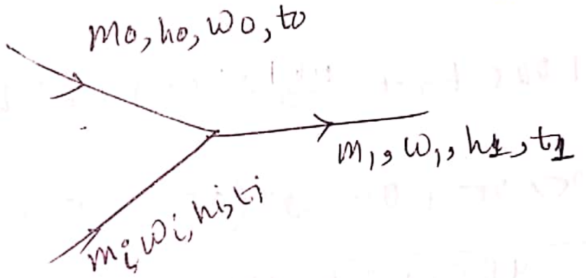
$\phi = \dfrac{P_{VA}}{P_{VSA}}$

$1 = \dfrac{P_{VA}}{0.014016}$

$\Rightarrow P_{VA} = 0.014016$ bar

$\omega_A = 0.00872$ kg/kg





$m = m_o + m_i = 500$ kg/s

$m_1 h_1 = m_o h_o + m_i h_i$

$h_1 = \dfrac{m_o h_o + m_i h_i}{m_1} = \dfrac{50 \times 92.17 + 450 \times 50.26}{500}$

$\Rightarrow$ $h_1 = 54.45$ kJ/kg

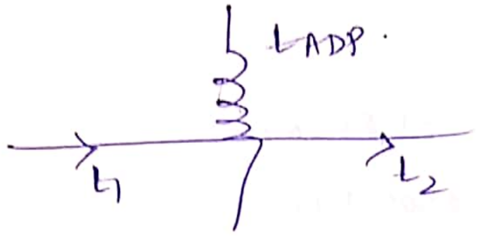
$t_1 = \dfrac{m_o t_o + m_i t_i}{m_1} = \dfrac{50 \times 45 + 450 \times 25}{500}$

$t_1 = 27°C$

$\omega_1 = \dfrac{m_o \omega_o + m_i \omega_i}{m_1} = \dfrac{50 \times 0.018 + 450 \times 0.00987}{500}$

$\omega_1 = 0.0106$ kg/kg of da

~~0.15 = ...~~ BPF $= \dfrac{t_2 - t_{ADP}}{t_1 - t_{ADP}}$



$$X = \frac{t_2 - t_{ADP}}{t_1 - t_{ADP}} = \frac{h_2 - h_{ADP}}{h_1 - h_{ADP}} = \frac{\omega_2 - \omega_{ADP}}{\omega_1 - \omega_{ADP}}$$

$$0.15 = \frac{t_2 - 12}{27 - 12} = \frac{h_2 - 34.06}{54.45 - 34.06} = \frac{\omega_2 - 0.00872}{0.0106 - \omega_{ADP}}$$

$\Rightarrow$ $t_2 = 14.25°C$ $h_2 = 37.1185$ KJ/kg $\omega_2 = 0.009$ Kg/Kg of da

$h_3 = 1.005 t_3 + \omega_3 (2500 + 1.88 t_3)$

$h_3 = 1.005 \times 25 + 0.00899 (2500 + 1.88 \times 25)$

$h_3 = 48.0225$ KJ/kg

at point - 3

$\omega_3 = \omega_2 = 0.00899$

$T_i = T_3 = 25°C$

$RSH = (h_3 - h_2) \dot{m}_1 = (48.0225 - 37.1185) \times 500 = 5452$

$RLH = \dot{m}_1 ((h_1 - h_3)) = (54.45 - 48.0225) \times 500 = 3213.75$

$G.L.C.C = (h_1 - h_2) \dot{m}_1 = (54.45 - 37.1185) \times 500 = 8665.75$ KJ

H_2O vap condensed $= (\omega_1 - \omega_2) \times \dot{m}_1 = 0.805$ kg/s

---

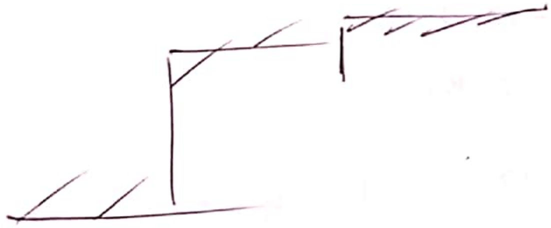
Q) 300 m^3/min of air at 10° DBT and 90% R.H is to be heated and humidified to 35° DBT & 22.5°C WBT. The reqd conds are achieved by heating, adiabatic humidification & then again by heating. The R.H of air coming out of humidifi is 90%. Find

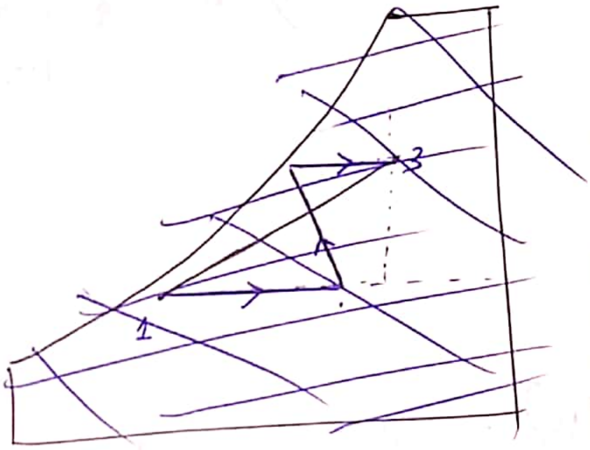
i) Heating capacity of I heater and the by-pass factor if the surface temp is 40°c.

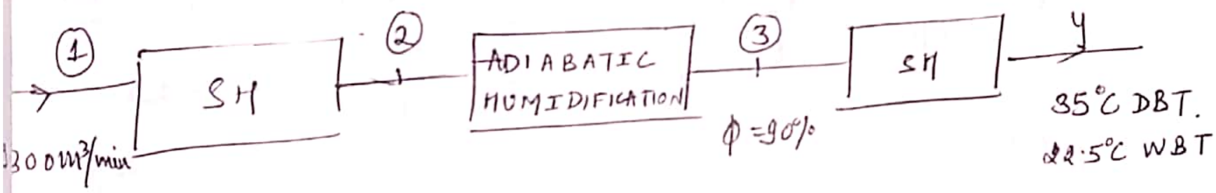
(ii) capacity of the humidifier in kg/hr.

(iii) Heating capacity of the II heater & the coil surface temp. if the by-pass factor is 50%.

(iv) Humidifying eff. of the air washer (humidifier)







DBT = 10°C
φ = 90°

From Psychrometric chart → $h_1 = 27$

$\omega_1 = 0.007$

$v_1 = 0.81\ m^3/kg$

$h_4 = 67\ KJ/kg$

$\omega_4 = 0.011$

$t_4 = 35°C$

$h_2 = h_3 = 48$

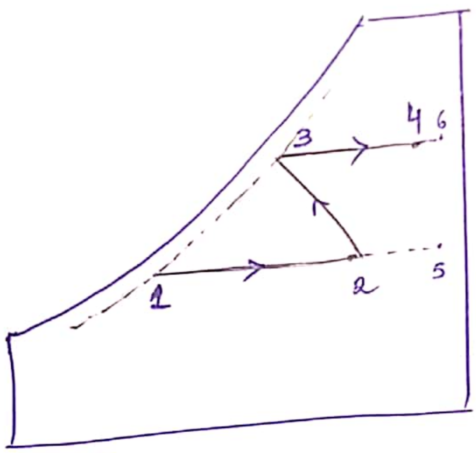
$\omega_2 = \omega_1 = 0.007$

$\omega_3 = \omega_4 = 0.011$

$t_2 = 32°C$

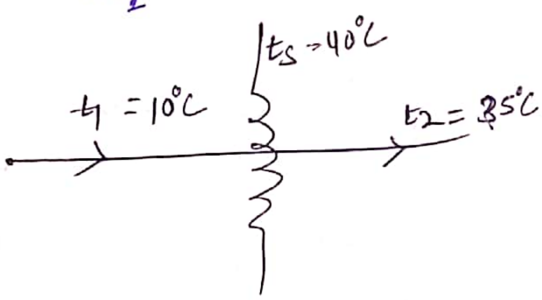
$t_4 = 35°C$

$t_3 = 18°C$



1) $(h_2 - h_1)\dot{m}_1 = (48 - 27) \times 6.17 = 129$ KW

$v_1 = \frac{vol}{m_1} \Rightarrow \dot{m}_1 = \frac{vol}{v_1} = \frac{(300/60)}{0.81} = 6.17$ kg/s



$BPF = \frac{40 - 35}{40 - 10}$

BPF = 0.26

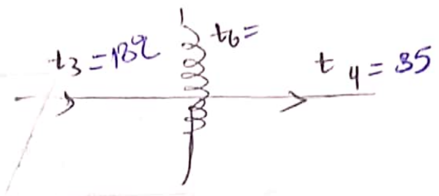
ii) $(\omega_3 - \omega_2)\dot{m}$

$= (0.011 - 0.007) \times 6.17 \times 3600$

$= 88.84$ kg/hr

iii) $(h_4 - h_3) \times \dot{m} = (67 - 48) \times 6.17$

$= 117.12$ kW



$t_6 > t_4 > t_3$.

$BPF = \dfrac{t_6 - t_4}{t_6 - t_3}$

$0.5 = \dfrac{t_6 - 35}{t_6 - 18} \Rightarrow \boxed{t_6 = 52°C}$

iv). $\eta = \dfrac{O/p}{I/p} = \dfrac{t_2 - t_3}{t_2 - t_3'} = \dfrac{32 - 18}{32 - 17} = 0.9333$

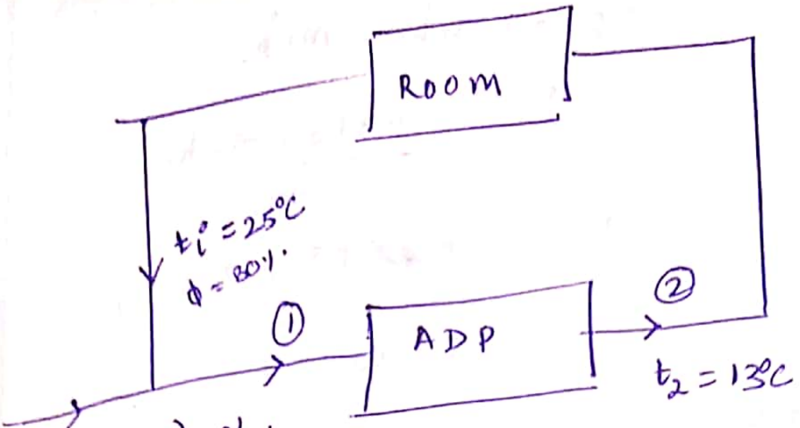
Q) An A/c plant supplies a total of 4500 m³/min of dry air which comprises 20% of fresh air at 40° DBT and 27° WBT and 80% of recirculated air at 25°C DBT & 50% R.H. Air leaves the cooling coil at 13°C saturated. Calculate the ~~total~~

i) Total cooling load

ii) Room heat gain

Sp. ~~heat~~ vol. of air at the entry of cooling coil. v = 0.869 m³/kg

Cond n	DBT(°C)	WBT(°C)	R.H %	w (gm/kg of da)	h (kJ/kg)
Outside	40	27	—	17.2	85
Inside	25	—	50	10.0	51
ADP	13	—	100	9.4	36.8

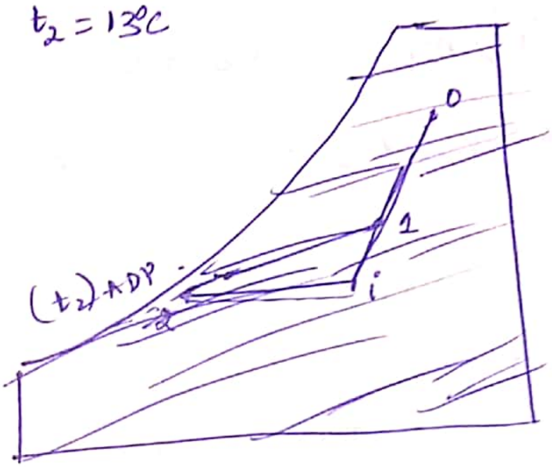


$v_i = (4500 \times 0.2)\ m^3/min$

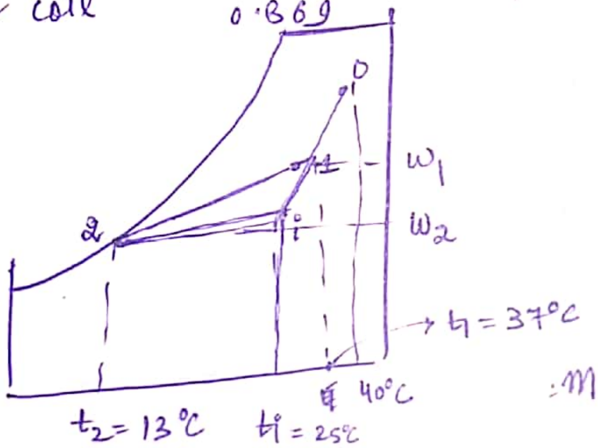
$m_i = \frac{(4500 \times 0.2)/60}{v_i}$ = ~~17.261 kg/s~~

DBT = 40°C

WBT = 27°C



mass entering cooling coil $= \frac{4500}{0.869} = 86.306$ kg/s



$P_V = P'_V - \frac{1.8\,p\,(t - t')}{2700}$

$P_V =$

$m_i t_i + m_o t_o = m_1 t_1$

$\Rightarrow t_1 = \frac{m_i t_i + m_o t_o}{m_1}$

$= \frac{0.8m \times 28 + 0.2m \times 40}{m}$

$\boxed{t_1 = 38°C}$

$\omega_1 = \frac{m_i \omega_i + m_o \omega_o}{m_1}$

$= \frac{0.2m \times 10.0 + 0.8m \times 17.2}{m}$

$\boxed{\omega_1 = 15.76 \text{ gm/kg of da}}$

i) $m_{H_2O\,cond} = (\omega_1 - \omega_2) \times \dot{m}_1$

$= (15.76 - 9.4) \times 86.306$

$= 0.5489$ kW

(i) $h_0 = 85$ KJ/kg

$h_2 = 36.8$ KJ/kg

$m_0 h_0 + m_i h_i = m_1 h_1$

$$h_1 = \frac{m_0 h_0 + m_i h_i}{m_1}$$

$$h_1 = 0.2 \times 85 + 0.8 \times 51$$

$$\boxed{h_1 = 57.81}$$

1) $(h_1 - h_2) m_1 = 1812$ kW

2) $(h_i - h_2) m_1 = 1225$ kW

NOTE: The degree of freedom for the moist air is 3.

$P + F = C + 2$

$1 + F = 2 + 2$

From Gibbs phase rule,

$P + F = C + 2 \Rightarrow 1 + F = 2 + 2 \Rightarrow F = 3$

(P = 1 as M.A remains in superheated state)

but on the psychrometric chart we found that with the help of any 2 variables we can locate or fix the state of any s/m bcoz the psychrometric chart is developed for atm pressure.

DOF: It is the min no. of independent (or intensive or intrinsic) variables required to locate or fix the state of any s/m.

Q) In an air conditioned room, the infiltration air is 3 air changes/hr. The temp diff b/w room and ambient is 20 K. Then the sensible heat load due to infiltration is.

Assume vol. of the room → 10 m^3.

30 m^3/hr

$\frac{30\ m^3}{60 \times 60}$

$= \frac{1}{120}$ m^3/s

~~$\frac{1}{120} \times 1000\ (20)$~~

$R_{SH} = 0.0204$ ~~[illegible]~~ $\times \Delta t$

$= 0.0204 \times cmm \times 20$

$3/hr = \frac{cmm}{10\ m^3}$

cmm ⇒ 30 m^3/hr

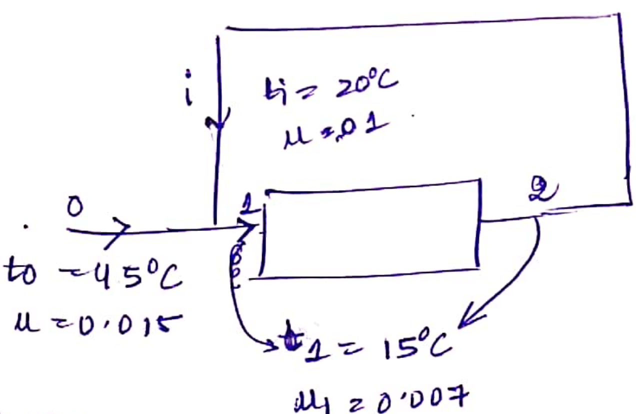
⇒ cmm = 0.5 m^3/min

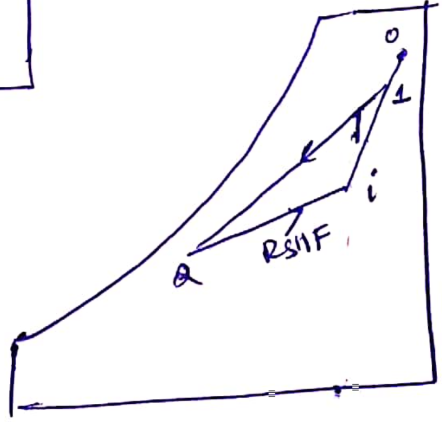
$R_{SH} = 0.0204 \times 0.5 \times 20$

$R_{SH} = 0.2\ kW$

Q) For an office building outdoor cond's are 45° DBT and humidity ratio of 0.015. The indoor design cond's are 20° DBT & 0.01 humidity ratio. The supply cond's are 15° DBT & 0.007 humidity ratio. If the supply air flow rate is 1000 m^3/min. Then the RSH & RLH are

a)
b)
c)
d)





no, use of ~~to~~ in calculating RSHF

$RSH = 0.0204 \times cmm \times \Delta t$

$= 0.0204 \times 1000 \times 10$

$\Rightarrow 204\ kW$

Scanned with CamScanner